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EXERGETIC ANALYSIS OF A COMBINED CYCLE GAS TURBINE USING DISTILLATE FUEL

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Abstract. *The present study aims to perform an exergetic analysis of each component of the combined cycle gas turbine (CCGT) using distillate fuel. Although it is of valuable information to the overall efficiency of the unit, overall exergetic efficiency presents no relevant data for an evaluation of improvement opportunities as only input values and Exergy cycle output are considered. The methodology consisted in collecting the data required for the preparation of models that are in accordance with literature on the subject, and calculating the efficiencies with EES software. The results demonstrate an exergetic efficiency of the plant on the order of 51%, and inform that the components that had the highest exergy destruction rates were the combustion chamber and the gas turbine. These results are in line with models presented by other authors. Finally, the analysis of the results enabled us to present suggestions for future improvements and studies.*

Keywords: *Exergy, Efficiency, Combined cycle, liquid fuel, mathematic model.*

1. INTRODUCTION

According to the national energy balance of 2013, approximately 76% of the electricity produced in Brazil comes from hydroelectric plants: a type of generation that is largely dependent on precipitation. In face of this difficulty, the government is investing in diversification of energy sources, in order to avoid the occurrence of another power rationing – as the one of 2001. Because of the importance of thermal energy in the present scenario, getting this energy in the most efficient manner possible is critical, due to the high cost of fuel used. So, an Exergy analysis of a thermal power plant was carried out. This plant, situated in Canoas (Rio Grande do Sul), is dual-fuel: runs on natural gas or distillate fuel – a type of light diesel oil, named FOET (fuel oil for electric turbine). Currently, due to natural gas supply restriction in the region, the operation is performed on FOET, so our paper will only approach the analysis for this fuel.

The aim of this study is to present an exergetic analysis of each component of the cycle in order to assess opportunities for improvement. Therefore, it will demonstrate the efficiency of individual components to allow a better assessment of present efficiencies' variations.

Some previous studies that have shown exergetic reviews of plants for electricity generation are mentioned below. These works are the theoretical foundations of this study.

Kalatalo (2004) conducted an Exergy analysis of a combined cycle of a plant located in Uruguaiana, RS, Brazil, whose characteristics are similar to the unit studied in this work, but that runs on natural gas. The author presented a global Exergy efficiency of the plant of approximately 57% combined with high-level values of efficiency of the components.

Ersayin and Özgener (2014), based on the first and second law of thermodynamics, conducted a performance review of a combined cycle of a power plant (Turkey) that uses natural gas as turbine fuel. In this case, the heat recovery boiler is of supercritical type, and is regarded as a single volume control in the study. Their results showed that the exergetic efficiency is higher in the compressor and low in the combustion chamber. It is also shown that high temperature of environment causes a proportional loss of exergetic efficiency of the combined cycle and decreases the net electric power produced by the plant.

Cassetti et. al. (2013) conducted a study using exergetic evaluation methods, economic evaluation and environmental risk applied to a gas turbine. Through mathematical models, they reached optimal operational values in these three perspectives. The methodology proposed is very useful for making economic and technical decisions.

Branco (2005) conducted a thermoeconomic analysis of a plant located in Mato Grosso do Sul that was operating on an open cycle. The study aimed at proposing various operational configurations of combined cycle, varying steam pressure levels in the Rankine cycle, and then compare the exergetic and exergoeconomic efficiency with the operation of an open cycle. The exergetic efficiency observed in the work (three-pressure level in the boiler) was of 51%.

In his work, Guirardi (2008) developed an analysis of the influence of the conditions of atmospheric air in the performance of gas turbines with inlet air-cooling system. The author created models for different cooling systems in compressor air intake: an evaporative model and another for absorption. For both exergy analysis was performed. The paper concludes that the evaporative cooling compressor inlet air is the most economically way to increase electricity production in gas turbines.

Turan and Aydin (2014) developed an exergoeconomic and exergetic analysis of an aero derivative model of a gas turbine, therefore only an analysis of a Brayton cycle. Their conclusions contain exergetic efficiency results of about 39%, indicate that the highest exergetic destruction rate was observed in the combustion chamber, and that the highest exergetic efficiency occurs in the high-pressure turbine.

Uberti (2014) carried out a simulation work of the same plant under analysis in this work in IPSEpro software in order to validate the simulations with different loads. The results indicate the best efficiency in combined cycle operation with natural gas fuel reaching a global efficiency value of 57%. With the same configuration approached in this paper, the author's results demonstrate rates of exergetic efficiency of 53,4%.

2. SYSTEM DISCRIPTION OF THE POWER PLANT

The power plant analyzed in this work was opened in 2002, initially operating only on simple cycle. In 2014, the work of implantation of combined cycle came to an end. The unit consists of a General Electric 7FA gas turbine with 170 MW of capacity (GE Catalog, 2015) operating with FOET and a Siemens SST700 / 900 steam turbine with 90 MW of capacity (Siemens Catalog, 2015).

The plant consists of the key equipment illustrated in Fig. 1: gas turbine; heat recovery boiler; steam turbine and condenser. The process consists in the suction of air into the gas turbine compressor at a compression ratio of approximately 15: 1 (GE Catalog, 2015). Then the air is directed to a combustion chamber where combustion reaction occurs along with fuel entry. The gases resulting form the combustion expand in the gas turbine and, at the end of the expansion, gases still have high energy, with an approximate temperature of 600 ° C. These gases pass through a heat recovery boiler that generates steam at three levels of pressure for expansion in the steam turbine. The steam turbine has a high-pressure section where the discharge steam is sent back to the boiler to be reheated along with the average-pressure vapor stream that feeds the steam turbine of the same level of pressure.

One of the characteristics of this plant is being operated at maximum capacity available most of the time. Because it is a source of generation that is only required by the National Electric System Operator when hydroelectric generation is on the edge, the requested generation uses all available capacity. Thus, the maximum load operation data will be used to calculate the exergy analysis.

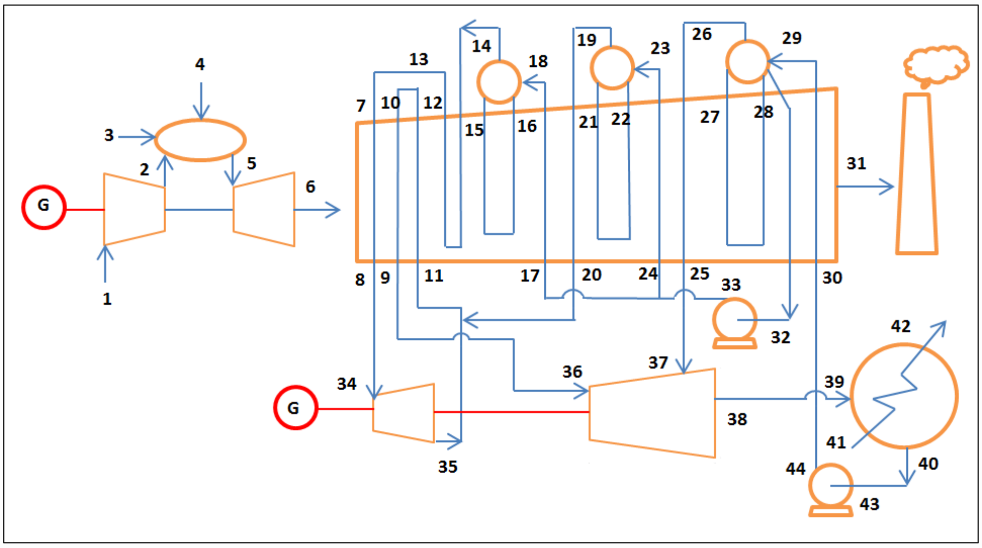


Figure 1. Schematic diagram of the analyzed plant.

The properties of temperature, pressure and mass flow rate of each point illustrated in Fig. 1 are described in Tab. 1 and were obtained directly from the supervisory system of the plant.

Table 1. Thermophysical properties of the plant points.

POINT	DESCRIPTION	FLUID	TEMPERATURE (°C)	PRESSURE (kPa)	MASS FLOW (kg/s)
1	Compressor inlet	air	14,5	101,3	431,7
2	Compressor outlet	air	400	1488	431,7
3	Combustion chamber demineralized water inlet	Water	14,5	3500	16
4	Combustion chamber Fuel input	FOET	30	3400	12,4
5	Combustion chamber outlet	CB	1276	1414	458,6
6	Gas turbine outlet	CB	600	102,8	458,6
7	Superheater HP-SH3 inlet	steam	473	10630	56,7
	Superheater HP-SH3 inlet	CB	594	167,4	458,7
8	Superheater HP-SH3 outlet	steam	550,2	10200	56,7
	Superheater HP-SH3 outlet	CB	545	160	458,7
9	Reheated steam RH2 outlet	steam	550,2	2220	62,2
	Reheated steam RH2 outlet	CB	545	155,2	458,7
10	Reheated steam RH2 inlet	steam	465	2430	62,2
	Reheated steam RH2 inlet	CB	590	162,3	458,7
11	Reheated steam RH1 inlet	steam	330,6	2360	62,2
	Reheated steam RH1 inlet	CB	545	152,7	458,7
12	Reheated steam RH1 outlet	steam	466	2360	62,2
	Reheated steam RH1 outlet	CB	447	146	458,7
13	Superheater HP-SH1/2 outlet	steam	476	10620	56,7
	Superheater HP-SH1/2 outlet	CB	447	141,6	458,7
14	Superheater HP-SH1/2 intlet	steam	320	10630	56,7
	Superheater HP-SH1/2 intlet	CB	545	157,5	458,7
15	Evaporator EVA HP outlet	steam/liquid	313	10620	56,7
	Evaporator EVA HP outlet	CB	326,7	137,4	458,7
16	Evaporator EVA HP intlet	steam/liquid	313,5	10600	56,7
	Evaporator EVA HP intlet	CB	447,7	143,7	458,7
17	Economizer ECO HP inlet	water / sat liq	137,2	14810	56,7
	Economizer ECO HP inlet	CB	326,7	135,2	458,7
18	Economizer ECO HP outlet	water / sat liq	312,5	14700	56,7
	Economizer ECO HP outlet	CB	153,9	111	458,7
19	Superheater SH IP inlet	steam	230	2830	8,8
	Superheater SH IP inlet	CB	327	135,2	458,7
20	Superheater SH IP outlet	steam	305	2500	8,8
	Superheater SH IP outlet	CB	255	129,3	458,7
21	Evaporator EVA IP outlet	steam/liquid	230	2830	8,8
	Evaporator EVA IP outlet	CB	205	118	458,7
22	Evaporator EVA IP inlet	steam/liquid	230	2830	8,8
	Evaporator EVA IP inlet	CB	255	123,4	458,7
23	Economizer ECO IP outlet	water / sat liq	182,5	4300	8,8
	Economizer ECO IP outlet	CB	151	107,7	458,7
24	Economizer ECO IP inlet	water / sat liq	134,4	4700	8,8
	Economizer ECO IP inlet	CB	205	112,6	458,7
25	Superheater SH LP outlet	steam	296	200	3
	Superheater SH LP outlet	CB	255	125,4	458,7
26	Superheater SH LP inlet	steam	148	200	3
	Superheater SH LP inlet	CB	315	131,2	458,7
27	Evaporator EVA LP outlet	steam/liquid	149	200	3
	Evaporator EVA LP outlet	CB	151	104,4	458,7
28	Evaporator EVA LP inlet	steam/liquid	148	200	3
	Evaporator EVA LP inlet	CB	205	109,3	458,7
29	Economizer ECO LP outlet	water / sat liq	149,4	1890	40,1
	Economizer ECO LP outlet	CB	120,3	101,6	458,7
30	Economizer ECO LP inlet	water / sat liq	67,9	2260	40,1

	Economizer ECO LP inlet	CB	151	106	458,7
31	Stack gases outlet	CB	120,3	101,6	458,7
32	High pressure pump inlet	water	134,6	490	63,6
33	High pressure pump outlet	water	137,6	15723	63,6
34	High pressure steam turbine inlet	steam	536	10010	53,4
35	High pressure steam turbine outlet	steam	336	2440	53,4
36	RH2 medium/low steam turbine inlet	steam	532	2140	62,2
37	Low pressure medium/low steam turbine inlet	steam	281	170	3
38	Medium/low pressure steam turbine outlet	steam	45	8,1	65
39	Condenser inlet	Saturated liquid	45	8,1	65
40	Condenser outlet	water	42	8,1	65
41	Condenser cool water inlet	water	30,7	269	3666,6
42	Condenser cool water outlet	water	42	240	3666,6
43	Condensed pump inlet	water	45	8,1	65
44	Condensed pump outlet	water	45,6	2440	65

Legend: CB: Combustion gases; HPSH3: high pressure superheater 3º pass; RH2: reheated steam 2º pass; HPSH1/2: high pressure superheater 2º e 1º pass; RH1: reheated steam 1º pass; EVAHP: high pressure evaporator; ECOHP1/4: high pressure economizer 1º to 4º passes; SHIP: medium pressure superheater; SHLP: low pressure superheater; EVAIP: medium pressure evaporator; ECOIP: medium pressure economizer; EVALP: low pressure evaporator; ECOLP: low pressure economizer.

3. EXERGETIC ANALISYS

Exergy is the energy that is available to be used. Exergy analysis is performed in the field of industrial engineering in order to employ energy more efficiently. The exergy, **X**, is characterized from the total system energy. Classifying the different types of associated Energy and Exergy that constitute the total energy according to the forms and peculiarities of transformation is convenient. As this paper discusses electricity production systems, only the modalities inherent to this process are going to be mentioned.

Besides, since the main objective of this research is to analyze thermal systems, the definition of work will be related to the concept of exergy and environment. Thus, work is approached here as a form of Energy that could be represented by the movement of a rotating shaft of a turbine, axial compressor, the action of a centrifugal pump, or the movement of the neighborhood of the system associated with environmental conditions (Lora, 2002).

The work called neighborhood work, W_{viz}, occurs when the system volume is changed against the environmental pressure until balance with the environment is reached. Considering any form of work in this system analysis as W and defining the work done against the neighborhood, Exergy is defined for the modality of Energy in the form of work X_{work} by Eq. (1):

$$X_{work} = W - W_{viz}. \tag{1}$$

Mass contains exergy, as well as proportional energy and entropy in a system. Similarly, the exergy, entropy and energy transport rates into or from the system are proportional to the mass flow (Çengel and Boles, 2013). When an amount of mass enters or exits a system, it carries an amount of exergy. The exergy of the mass flow X_{mass} is expressed by

$$X_{mass} = m\,(h - h_0) - T_0(s - s_0) + \frac{V^2}{2} + gz \tag{2}$$

Where m refers to the mass flow rate [kg / s], h represents enthalpy [kJ / kg], s is the entropy [kJ / kg.K,] v is the speed [m / s], T is the temperature [K], g is the gravity acceleration [m / s²], and z is the height [m]. In Eq. (2) the terms subscript "0" are taken in the ambient temperature. The following relation can represent Exergy destruction:
Eq. (3):

$$X_{destruction} = T_0 S_{ger} \tag{3}$$

thus, S_{ger} is the entropy generated by the internal irreversibilities. The exergy always decreases during a process in an isolated system, or remains constant in the extreme case of a reversible process. Therefore, the exergy never increases and in a real process it will always have a portion destroyed. In thermal systems like the one being discussed in this paper, it is common to analyze the fuel solely by a mixture of perfect gases and all system components are present in the environment. This consideration being made, one may adopt the method of Bejan et.al. (1996), which regards the chemical potential of each substance under system and environmental conditions. Thus the chemical Exergy **x_{quim}** can be expressed by

$$x_{quimica} = \sum \bar{g}_r(T_0, p_0)y_i - \sum \bar{g}_r(T_0, p_0)y_{i0} + \bar{R}T_0 \sum y_i ln\left(\frac{y_i}{y_{i0}}\right) \quad (4)$$

In Eq. (4), $\boldsymbol{g_r}$ is the Gibbs function of the specific substance (in molar basis), the temperature is $\boldsymbol{T}$ and the pressure is $\boldsymbol{P}$, specified by the summation where i = 0, 1, 2 ... In this case, when i = 0, it refers to environment conditions' values. The constant $\boldsymbol{R}$ is the universal constant for an ideal gas and $\boldsymbol{y_i}$ is the molar fraction of the substance $\boldsymbol{i}$. Considering that chemical reactions occur in an isothermally and reversibly way, the corresponding exergy is expected to be approximately equal to the change in Gibbs function during reaction, baring in mind that the third term of the equation above is much smaller than the other ones. In reactions involving fuels composed of hydrocarbons, the difference between the variation in Gibbs function and the lower heating value (LHV) is very small. Thus, the lower calorific value is usually admitted for chemical exergy of the fuel.

As the exergy balance was conducted in a control volume, power flows associated with mass flows crossing the control surface were also considered. Energy conservation for a control volume is given by Eq. (5) (Bejan, 1996).

$$\frac{dX_{vc}}{dt} = \sum_j \left(1 - \frac{T_0}{T_f}\right) \dot{Q}_j - \left(\dot{W}_{vc} - p_0 \frac{dV_{vc}}{dt}\right) + \sum \dot{X}_{mi} - \sum \dot{X}_{mo} - \dot{X}_d, \quad (5)$$

The sub-index $\boldsymbol{j}$ represents the amount of heat flows, $\boldsymbol{Q_j}$, present between the boundary at ambient temperature $\boldsymbol{T_0}$ and the control volume. The second term of the equation refer to workflows that cross the control volume. The summations $\boldsymbol{\Sigma Xmi}$ and $\boldsymbol{\Sigma Xmo}$ represent the exergy flows associated with mass flows that enter and exit the control volume, respectively.

The majority of control volumes found in practice, including the ones approached in this paper, operate in steady state. This is the case of turbines, compressors, nozzles, diffusers, pumps and heat exchangers that do not suffer variations in mass quantities, volume, energy, entropy and exergy this way. Therefore, the amount of exergy (heat, work, mass flow) that enters a system with steady state flow must equal the amount of exergy leaving plus the amount of exergy destroyed (Çengel and Boles, 2013). That being so, the expression for the overall rate of exergy balance for a process in continuous flow is reduced to Eq. (6):

$$\sum_j \left(1 - \frac{T_0}{T_f}\right) \dot{Q}_j - \dot{W} + \sum \dot{X}_{mi} - \sum \dot{X}_{mo} - \dot{X}_d = 0 \quad (6)$$

For control volume operating at steady state, the amount of exergy that leaves the control volume is always lower than the admitted and the difference between them is associated with exergy destruction due to the irreversibilities of the process (Kotas, 1995). The set of equations that express the exergy balance in the control volume was developed in this manner. These equations were used to develop the exergetic analysis of each component of the power generation cycle. In the special case of the combustion chamber the approach also included chemical exergy (chemical reactions).

4. METHODOLOGY

Efficiency was determined in a way that meets the peculiarities of every equipment involved. The methodology employed for this assessment was proposed by Bejan et. al. (1996), which conveniently classifies exergy flows that enter and exit the control volume. In this way, an analysis criterion was obtained, allowing quantitative determination of what portions of contribution to the overall value of system efficiency are associated with each component, as well as of the global value of the system efficiency. Exergetic efficiency can therefore be expressed by Eq. (7),

$$\varepsilon_{ex} = \frac{\text{exergy of products}}{\text{exergy of the resources used}} \quad (7)$$

To calculate the efficiency of each component a model using the software EES (Engineering Equation Solver) was developed. The Fig. 2 illustrates the summary diagram of the conventions adopted to calculate the efficiency of each component present on the cycle. The input data for models such as pressures, temperatures and flow rates were obtained directly obtained from processes in the plant (supervisory system). The internal pressures of the boiler (next gases) were calculated from indication of Bejan et. al.(1996) and Kalatalo (2004).

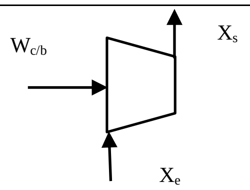
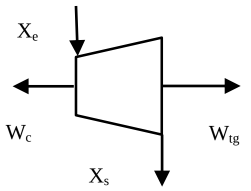
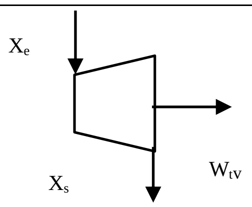
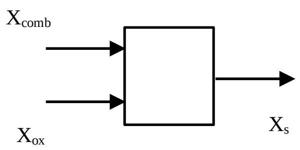
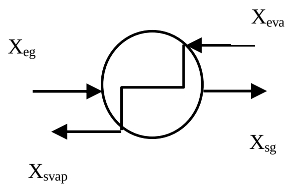
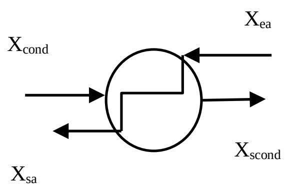
Equipment	Graphic representation of de control volume	Exergetic efficiency
Compressor or Pump		$\varepsilon_{c/b} = \frac{X_s - X_e}{W_{c/b}}$ (8) $\varepsilon_{c/b}$ = Compressor or pump efficiency X_s = Outlet exergy [kW] X_e = Inlet exergy [kW] $W_{c/b}$ = Compressor or pump work [kW]
Gas turbine		$\varepsilon_{tg} = \frac{W_{tg} - W_c}{X_e - X_s}$ (9) ε_{tg} = Gas turbine efficiency X_s = Outlet exergy [kW] X_e = Inlet exergy [kW] W_{tg} = Gas turbine work [kW] W_c = Compressor work [kW]
Steam turbine		$\varepsilon_{tv} = \frac{W_{tv}}{X_e - X_s}$ (10) ε_{tv} = Steam turbine efficiency X_s = Outlet exergy [kW] X_e = Inlet exergy [kW] W_{tv} = Steam turbine work [kW]
Combustion chamber		$\varepsilon_{cb} = \frac{X_s}{X_{comb} + X_{ox}}$ (11) ε_{cb} = Combustion chamber efficiency X_{comb} = Fuel exergy [kW] X_{ox} = Oxidant exergy [kW] X_s = Outlet products exergy [kW]
Heat exchanger HRSG		$\varepsilon_{tc} = \frac{X_{svap} - X_{evap}}{X_{eg} - X_{sg}}$ (12) ε_{tg} = Heat exchanger HRSG efficiency X_{svap} = outlet steam exergy [kW] X_{evap} = Inlet steam exergy [kW] X_{eg} = Inlet gases exergy [kW] X_{sg} = Outlet gases exergy [kW]
Heat exchanger Condenser		$\varepsilon_{cond} = \frac{X_{scond} - X_{econd}}{X_{ea} + X_{sa}}$ (13) ε_{tg} = Condenser efficiency X_{econd} = condensate inlet exergy [kW] X_{scond} = Condensate outlet exergy [kW] X_{ea} = Water inlet exergy [kW] X_{sa} = Water outlet exergy [kW]

Figure 2. Summary of the conventions adopted for each type of component of cycle component.

5. RESULTS

Tab. 2 illustrates the results of exergy rate of resources and products and the exergetic efficiency found in the application of models with the input data listed in Tab. 1, in the ESS software.

The results in Tab. 2 show that the components of lower exergetic efficiency are the boiler heat exchangers and the superheaters (SHIP, SHLP and EVALP). It is import to notice that the components from TC-SH3 to TC-ECOLP represent a single component of the combined cycle, in this case, the heat-recovering boiler. The overall exergetic efficiency of this component is on the order of 49%.

Table 2. Results of exergetic evaluations by component; for combined cycle operating with FOET.

Component	Exergetic Efficiency	Descruction Exergy Rate per Component [kW]	Percentage destruction per component	Percentage destruction relative by inlet Exergy
Compressor	0,93	5456,43	0,0236	0,01155
Comb. chamber	0,73	73331,58	0,3169	0,15526
Gas turbine	0,56	78178,54	0,3378	0,16552
TC - SH3	0,37	5137,08	0,0222	0,01088
TC - RH2	0,37	4895,24	0,0212	0,01036
TC-SH1/2	0,47	8492,26	0,0367	0,01798
TC - RH1	0,31	10459,30	0,0452	0,02215
TC- EVAHP	0,98	204,19	0,0009	0,00043
TC- SHIP	0,24	10792,82	0,0466	0,02285
TC- SHLP	0,20	723,30	0,0031	0,00153
TC- EVAIP	0,50	2788,67	0,0120	0,00590
TC - ECOIP	0,31	588,65	0,0025	0,00125
TC - EVALP	0,18	4221,10	0,0182	0,00894
TC- ECOLP	0,59	1902,38	0,0082	0,00403
HP steam turbine	0,93	630,29	0,0027	0,00133
IP/LP steam turbine	0,81	6082,73	0,0263	0,01288
Condenser	0,94	422,11	0,0018	0,00089
Condensed pump	0,84	13,29	0,0001	0,00003
High pressure pump	0,93	36,32	0,0002	0,00008
Legend: HPSH3: high press. superheater 3º pass; RH2: reheated steam 2º pass; HPSH1/2: high press. superheater 2º e 1º pass; RH1: reheated steam 1º pass; EVAHP: high press. evaporator; ECOHP1/4: high press. economizer 1º to 4º passes; SHIP: medium press. superheater; SHLP: low press. superheater; EVAIP: medium press. evaporator; ECOIP: medium press. economizer; EVALP: low press. evaporator; ECOLP: low press. economizer.				

Evaluating the results of Tab. 2, it is also possible to observe that despite very low efficiencies in heat exchangers of the heat-recovery boiler, they have very little influence on the total exergy destruction provided for the system. This is due to the fact that the greater contribution to the destruction of exergy occurs in the upper cycle, that is, in components such as the combustion chamber and the gas turbine.

The combustion chamber and the gas turbine showed the highest exergy destruction of the system, representing 15% and 16% of the total exergy supplied. Kalatalo (2004) also obtained this finding in the study of Uruguaiana power plant in which he also found low efficiencies on the boiler heat exchanger. Ersayin and Özgener (2014) also found that the highest exergy destruction rate was in the combustion chamber.

In Tab. 3, the results of the thermal efficiencies calculated for the open cycle and combined cycle are illustrated. These results match to the two operating options in plants of combined cycle.

Table 3. Results of thermal and exergetic efficiency of the system.

	Open cycle (Brayton)	Combined cycle (Brayton + Rankine)
Thermal efficiency (1ª law)	0,36	0,53
Exergetic efficiency calculated by models (2ª law)	0,35	0,51

In the results of Tab. 3, a considerable improvement in operating efficiency in combined cycle in relation to open cycle can be observed, in accordance with the results obtained by Branco (2005). The author found 51% to a simulation with three steam pressure levels. The efficiency found by Uberti (2014) was 3.4% higher than the efficiency shown in this work, but it should be considered that the models used were based on design data, and that the software used (IPSEpro) has some restrictions on the model of the combustion chamber. Turan and Aydin (2014) found an efficiency of second law of approximately 39% in the Brayton cycle of a gas turbine of aeroderivative type, which is very close to the result obtained for the turbine evaluated in this study. It is known that the aeroderivative turbine model has more efficiency than the one analyzed in our work, which is a "heavy-duty" gas turbine.

Table 4. Influence of exergy destruction on each cycle of the system.

Cycle	Participation on the rate of exergy destruction of the system
Brayton (superior)	0,68
Rankine (inferior)	0,32

Tab. 4 illustrates the contribution of each cycle in the destruction of exergy supplied to the system. Since components have the highest exergy destruction rates in Brayton cycle, the result reflects on the low exergetic efficiency of this cycle.

6. CONCLUSIONS

The Exergy analysis applied in this study proved to be a valuable tool for glimpsing possibilities of improvements in plant efficiency. Therefore, it is possible to concentrate efforts on the optimization of equipment and processes that will really have impact on the overall efficiency of the system, i.e. the combustion chamber and gas turbine.

Considering these results, many possibilities for future studies arise, including: exergoeconomic analysis of the plant, studies focused in equipment with greater exergy destruction rate (i.e. combustion and turbine gas chamber). Other possibilities refer to further study of the feasibility of increasing the generation of electrical work through implantation of a cooling system for the inlet air of the compressor, as well as the viability of changing water of injection for steam in the combustion chamber.

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